TECHNICAL REPORT

Project E-100-344

SOLID WASTE CYCLONE INCINERATOR

By

J. F. Kinney and J. M. Akridge

January 1969

Engineering Experiment Station
GEORGIA INSTITUTE OF TECHNOLOGY
Atlanta, Georgia

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I. SUMMARY

In January 1968, a program was initiated to study a cyclone type incinerator as a possible means of efficiently burning peanut hulls without polluting the air.

Following a theoretical study of the separation possibility of small particles in a cyclone-type burner, a conical model was constructed to study advection. Despite difficulties in obtaining the desired flow field, the unit produced separation of ground peanut hulls with hot gases (1000° F) produced by burning Atlanta City gas. A cylindrical model was also fabricated from clear plastic using a contoured bottom plug, which not only produced a very high speed rotational inner vortex but the contoured plug produced a turbulent "mixing" or "turnover" zone where the retention time believed to be required to burn peanut hulls to ash may be practical.

It is concluded that:

1. Natural advection of secondary air is not practical in the model tested and probably not in any model because of the high centrifugal pressure.

2. Forced advection of secondary air in the model tends to encourage short-circuiting of the cyclonic flow out the stack in the model tested.

3. The higher temperature creating less dense gases and a higher gas viscosity does not appear to be detrimental - at least in the range to 1300° F.

4. Peanut hulls are readily separated in hot gases obtained from burning Atlanta City gas with air.

5. Residence times greater than 1/10 second are required to char peanut hulls in the model (1 ft. diameter) cyclone 'burner' with 1000° F gases. Burning to ash will require longer residence time and probably higher temperature.

6. A cylindrical cyclone with contoured bottom plug may be better adapted for this type service.

7. A turbulent "turnover compartment" may be the compromise to burning materials such as peanut hulls which require a significant residence time to ash and still maintain velocities to insure particulate removal from the exhaust gases.

It is recommended that:

1. The next phase should include two model "burners", a cylindrical burner with contoured plug, and a conical burner with more conventional proportions and a contoured plug.

2. The burners be constructed without secondary air ports initially.

3. Work with a large 'burner' be delayed until the preliminary work in 1 (above) is completed to determine which type burner should be constructed.

II. PURPOSE

1. The objective of this research is to make a theoretical analysis of the gas flow in a cyclone type incinerator designed to burn low density solids.

2. After a theoretical analysis has been made, a model will be built to check the accuracy of the analysis. The model will be unfired and used primarily to check the gas flow only.

3. This study is expected to provide background data for the design of an incinerator of useable size.

III. BACKGROUND

Because of atmospheric pollution, the disposal of solid wastes of all types has become a major problem throughout the nation. Increases in the quantity of waste, and the high cost of land and transportation, have made on-site burning more and more attractive.

Unfortunately, burning only changes the form of the problem with the incinerators presently being used at many locations. Federal and state agencies are rapidly moving to enforce strict requirements concerning atmospheric pollution resulting from burning waste materials. These requirements are proving difficult to meet with certain types of wastes.

In the Southeastern region of the United States low density wastes from peanuts and cotton pose a difficult problem. The expanding peanut processing industry finds itself with 157,000 tons of peanut hulls annually with a bulk density of five pounds per cubic foot. Attempts at disposing of these hulls by other means have been unsuccessful to date.

The Peanut Shellers Association asked Georgia Tech for assistance with the problem. In January 1968 a program was initiated to study a cyclone type incinerator as a possible means of efficiently burning the hulls without polluting the air. Dr. Charles Bouchillon of Mississippi State University, acting as a consultant to the Mechanical and Industrial Sciences Branch of the Georgia Tech Engineering Experiment Station, conducted a theoretical analysis of a cyclone burner design. This study was based on the data listed in Table I - which was the best information available from several sources, including the Peanut Shellers Association.

1. Quantity of material to be burned - 2000- 8000lbm/hr.
2. Physical characteristics of the material
    a. Initial particle size - 50μ - 3/8 x 3/8 - 3/8 x 1.0
    b. Surface area/weight - 10 in^2/gm
    c. Specific gravity - 0.2 - 0.25
    d. Heating value - 9000 Btu/lbm
    e. Ash content - 2.5%
    f. Softening temperature of ash - 2100° F
    g. Fluid temperature of residue - 2150° F
3. Required separation effectiveness - 85 - 95%
4. Energy for forced ventillation - economically justifiable
5. Discharge - to atmosphere
6. Air to fuel ratio - 16 to 1 (160% excess air)

TABLE I

BASIC DATA FOR PEANUT HULL CYCLONE BURNER ANALYSIS

One of the major problems anticipated with burning peanut shells in a vortex-type burner is associated with the small amount of sand that cannot be separated from the shells. This sand, impinging on the walls of the burner would creat an erosion problem. In order to defect the sand, or at least reduce its impact energy, a curtain of air next to the burner wall could be admitted to the burner through a number of ports, the advection ports. This had other advantages, 1) all of the air need not be admitted at the top, 2) the advected air would tend to keep the walls cool and thus decrease erosion, and 3) decrease the formation of oxides of nitrogen by operating at lower temperatures.

Free advection by the high velocity of the circulating gases in the vortex would be a neat trick, if practical. If not practical, air could be forced in from a manifold, at some increase in initial and operating costs.

Figure 1 shows the basic design proposed by Bouchillon. The design, of course, was based on a number of assumptions since very little data is available on vortex burners. The design was directed primarily at good performance as a separator, i.e., few particles would escape from the exhaust stack whether they burned or not. Table II shows the theoretical equilibrium particle diameter for several radii and particle densities; this is the particle size for marginal separation, larger sizes would be readily separated from the exhaust gases.

The results of Bouchillon's analysis were encouraging. This preliminary study indicated that it was technically feasible to build a vortex incinerator with an emissions output considerably below state and federal requirements.*

A second major problem associated with the vortex burner is the combustion of light-weight material. Particulate matter requires a finite time to burn. The process of burning not only adds energy to the vortex in the form of heat but also adds additional gaseous material to the vortex. These problems were not covered in the preliminary work.

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*Dr. Bouchillon's report is included as Appendix A.

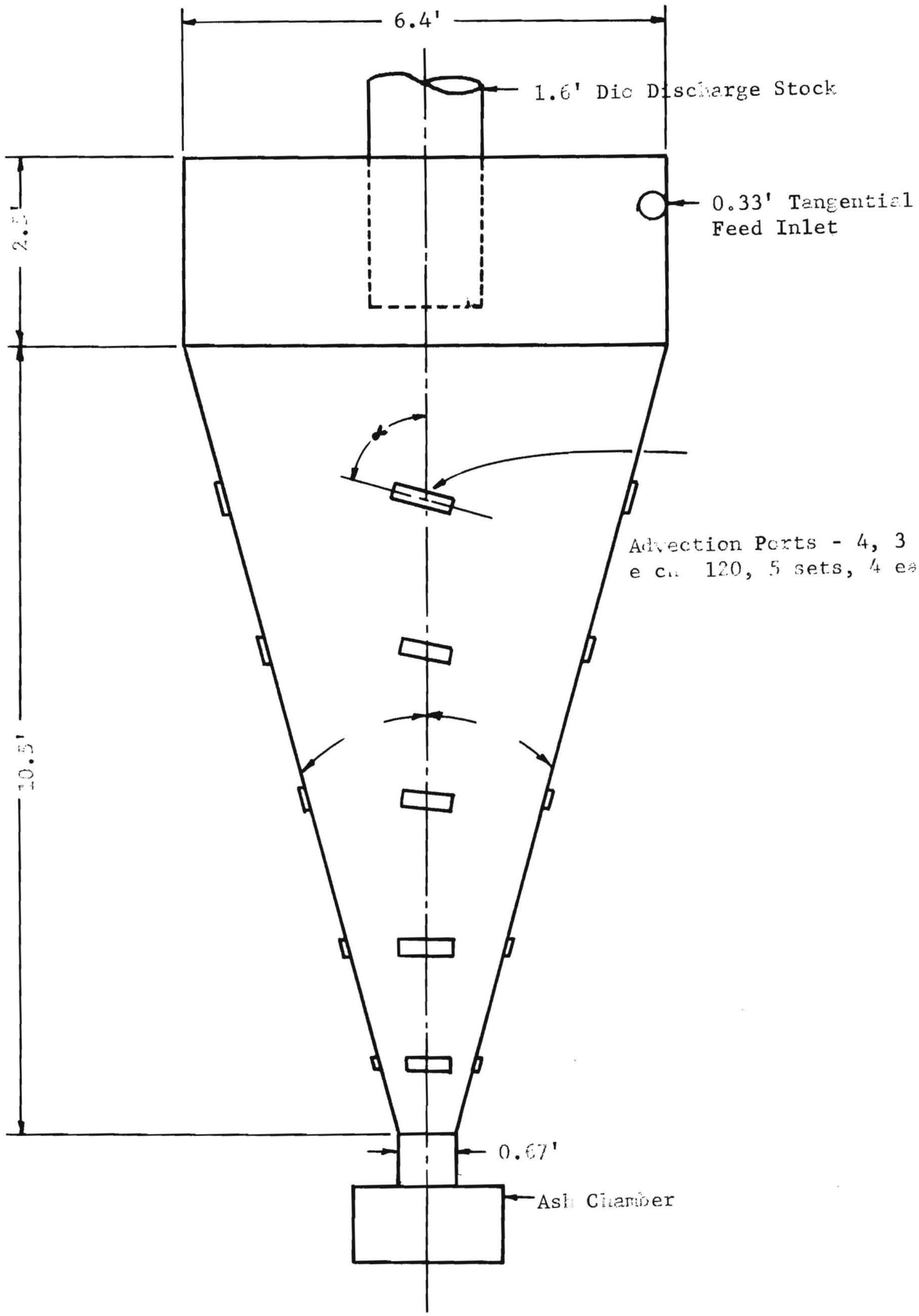


Figure 1. Sketch of Proposed Free Convection Driven Vortex burner.

Density $\frac{lbm}{ft^3}$	Radius ft	Particle Size Microns	Reynolds Number
Ps	R	A	Re
32.00	3.20	7.32	0.175
32.00	2.80	6.27	0.160
32.00	2.40	5.23	0.142
32.00	2.00	4.19	0.122
32.00	1.60	3.17	0.100
64.00	3.20	5.17	0.124
64.00	2.80	4.43	0.113
64.00	2.40	3.69	0.100
64.00	2.00	2.96	0.086
64.00	1.60	2.24	0.070
96.00	3.20	4.22	0.101
96.00	2.80	3.62	0.092
96.00	2.40	3.01	0.082
96.00	2.00	2.42	0.070
96.00	1.60	1.83	0.057
128.00	3.20	3.65	0.087
128.00	2.80	3.13	0.079
128.00	2.40	2.61	0.071
128.00	2.00	2.09	0.061
128.00	1.60	1.58	0.050

TABLE II

EQUILIBRIUM PARTICLE SIZES FOR VARIOUS PARTICLE DENSITIES AND RADIAL POSITIONS IN THE POSTULATED FLOW FIELD

IV. EXPERIMENTAL WORK

A. Conical model

Work by several investigators has indicated that scaling cyclone separators is exceedingly difficult if not impossible. However, a model of the design considered in the theoretical analysis could answer several questions even if the result could not be scaled up to a full size incinerator. With this in mind a model (approximately 1/6 scale) was built and tested. Figure 2 shows the initial design and Figure 3 shows the detail of the port attachment. Figure 4 is a picture of the initial model.

The model was equipped with an ac-dc motor blower capable of supplying air at velocities from zero to above 200 feet/second. The motor was connected to a variable voltage supply so that motor speed could be adjusted as desired.

B. Air tests

The model in this configuration was used to determine whether the advection ports would aspirate the secondary air. It was quickly realized that the centrifugal head on the interior of the burner was greater than atmospheric: flow through the advection ports was from inside to outside at all primary air velocities. With approximately 120 ft/sec inlet air, the following static pressures were measured at the port numbers indicated in Figure 2, (with all other ports plugged).

Port No.	Static Pressure, in H_2O
4	3.8
3	2.5
2	2.3
1	1.5

An approximate analysis of the centrifugal pressure magnitude is shown in Appendix B.

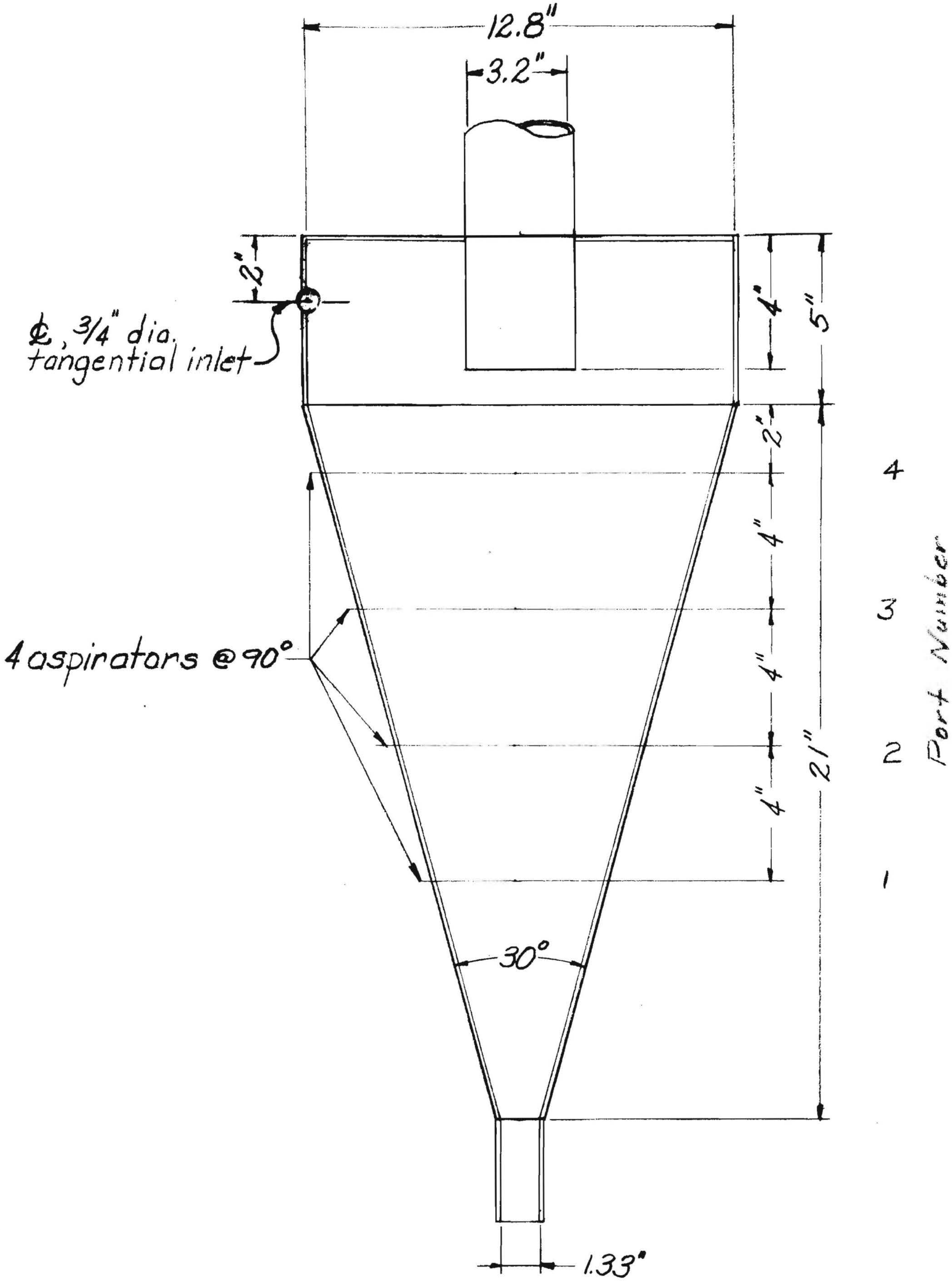


Figure 2. Dimensions of Conical Model

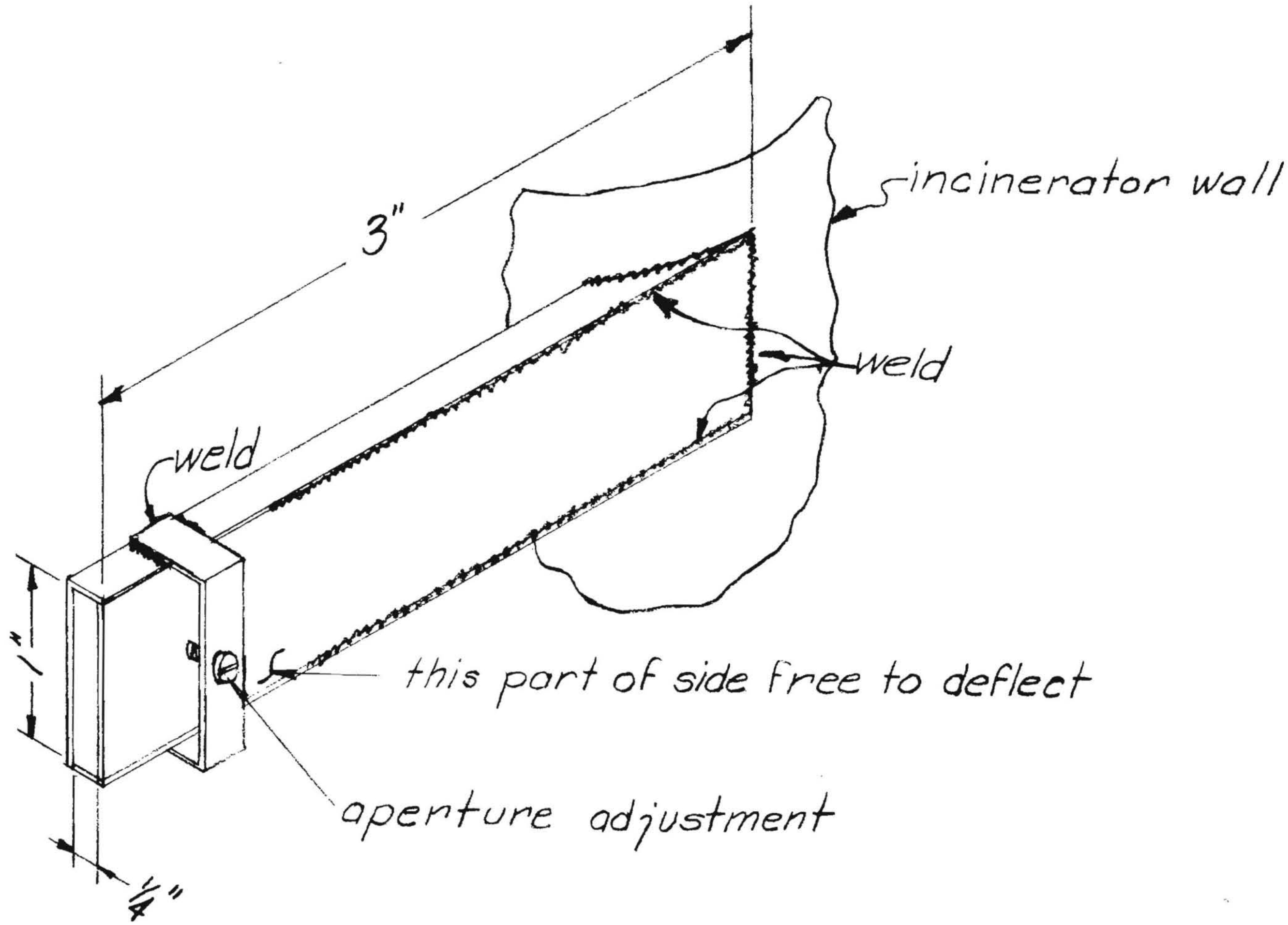


Figure 3. Detail of Advection Port and Attachment.

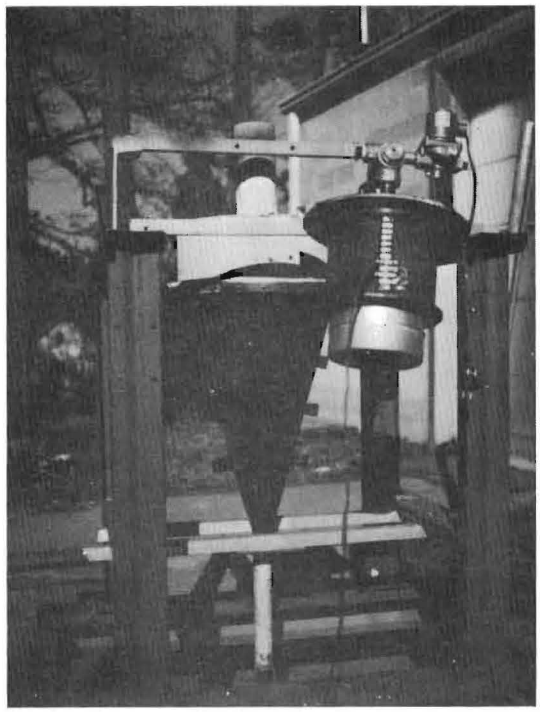


Figure 4. Initial Conical Model

Figures 5 and 6 show the model adapted for forced secondary air. The auxiliary air inlets were encased in a can and the can pressurized with a second ac-dc motor connected to a variable voltage source. The quantity of auxillary air could be varied over a wide range by varying the motor speed.

C. Burner tests

Initially, the model 'burner' was intended only to verify natural advection. However, since the hardware was on hand with the exception of a gas-air mixing valve and flame holder, cursory testing with hot gases as contrasted with cold gases appeared worth-while. This is particularly true since the viscosity of a gas increases with temperature and since the density of the gas varies inversely with temperature. And, as the temperature increases the uniformity of temperature within the 'burner' should decrease, due to distributed heat losses. Consequently, cursory testing with a hot gas obtained by burning Atlanta City gas with air was attempted.

For the tests using Atlanta City gas, the 'burner' was modified by feeding the original blower output to a North American gas-air mixing valve and then directing the unburned mixture tangentially into the model. This input is referred to as the primary air. A stainless steel flame holder was attached to the inside of the model at the primary air inlet. A spark-plug connected to a high voltage supply was used to ignite the gas air mixture adjacent to the flame holder. Figure 7 shows the flame holder and inlet design.*

The model was equipped for this series of tests with thermocouples projecting 0.125" from the inside surface and located just before and just after

*The square patches in the picture are plates welded over the advection ports later in the program when the picture was taken.

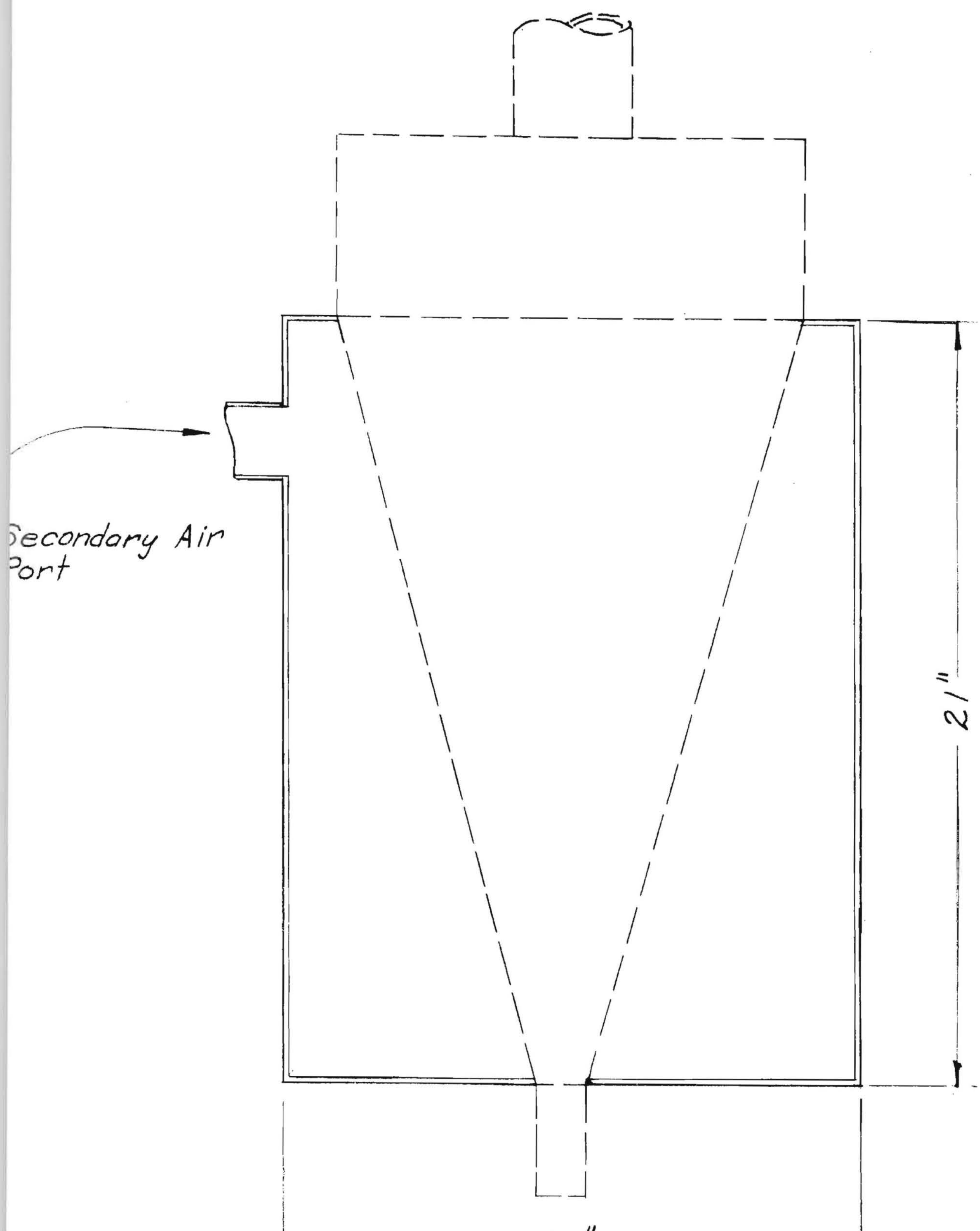


Figure 5. Modification for forced convection.

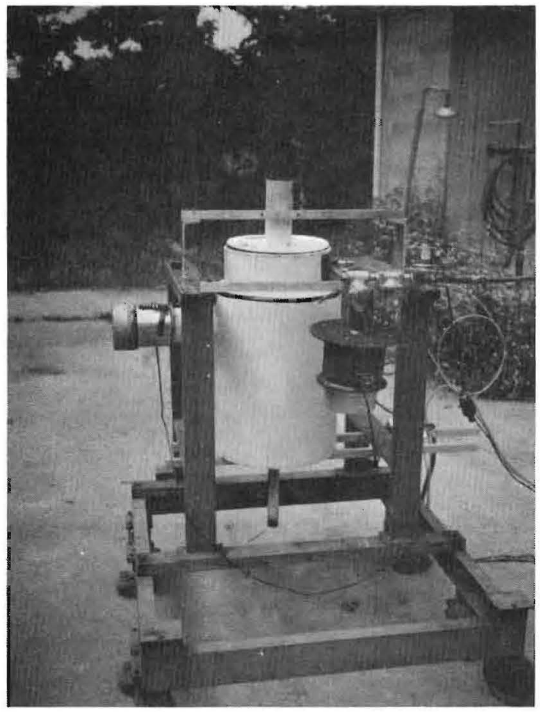


Figure 6. Model Incinerator with Forced Secondary Air

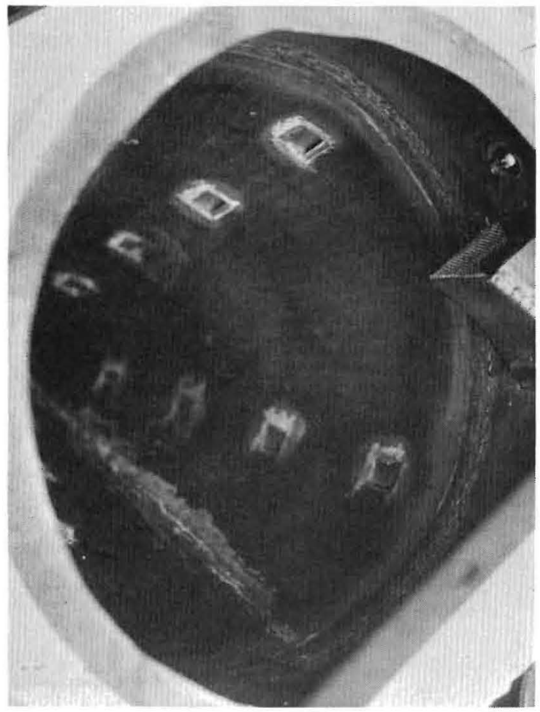


Figure 7. Picture of Primary Air Inlet with Flame Holder.

one port at each of the four levels. The exhaust stack was also equipped with a thermocouple. The exhaust thermocouple measured the exhaust temperature 0.125" from the stack wall.

The exhaust stack extended four (4) inches below the top of the incinerator, i.e., extended 4" into the incinerator for this series of tests. Temperature distribution along the walls reached a maximum of 600° F at a point just before the first auxillary air inlet. The temperature decreased in the lower regions of the chamber. This was initially believed to reflect the addition of auxillary air. Later tests showed this to be only partially true. Figure 8 shows the temperature distribution along the burner wall, together with a sketch showing the relationship between the thermocouples and advection ports.

Apparent lack of flow in the lower regions of the incinerator prompted a change in the air-fuel inlet direction from horizontal to 5° downward. Pitot probes were added at several points in an attempt to determine the exact flow direction. No measureable change was noted from the previous tests and no measurable velocities were detected by the pitot probes.

A moveable temperature probe was added through the solid particle discharge port. This probe could be raised vertically along the burner axis to a point 2" below the bottom of the exhaust stack. At no point was a temperature measured along the wall. The exhaust temperature during all of these tests was approximately 1000° F higher than that measured at any other point.

It was apparent at this point that the primary flow was going only as low as the bottom of the exhaust stack and was then short-circuiting out the stack. A thermocouple probe was added to the top of the incinerator at an angle equal to the slope of the burner cone. This thermocouple was used to traverse from the top to the bottom approximately two (2) inches from the burner wall. Figure 9 shows the location of this probe in the burner.

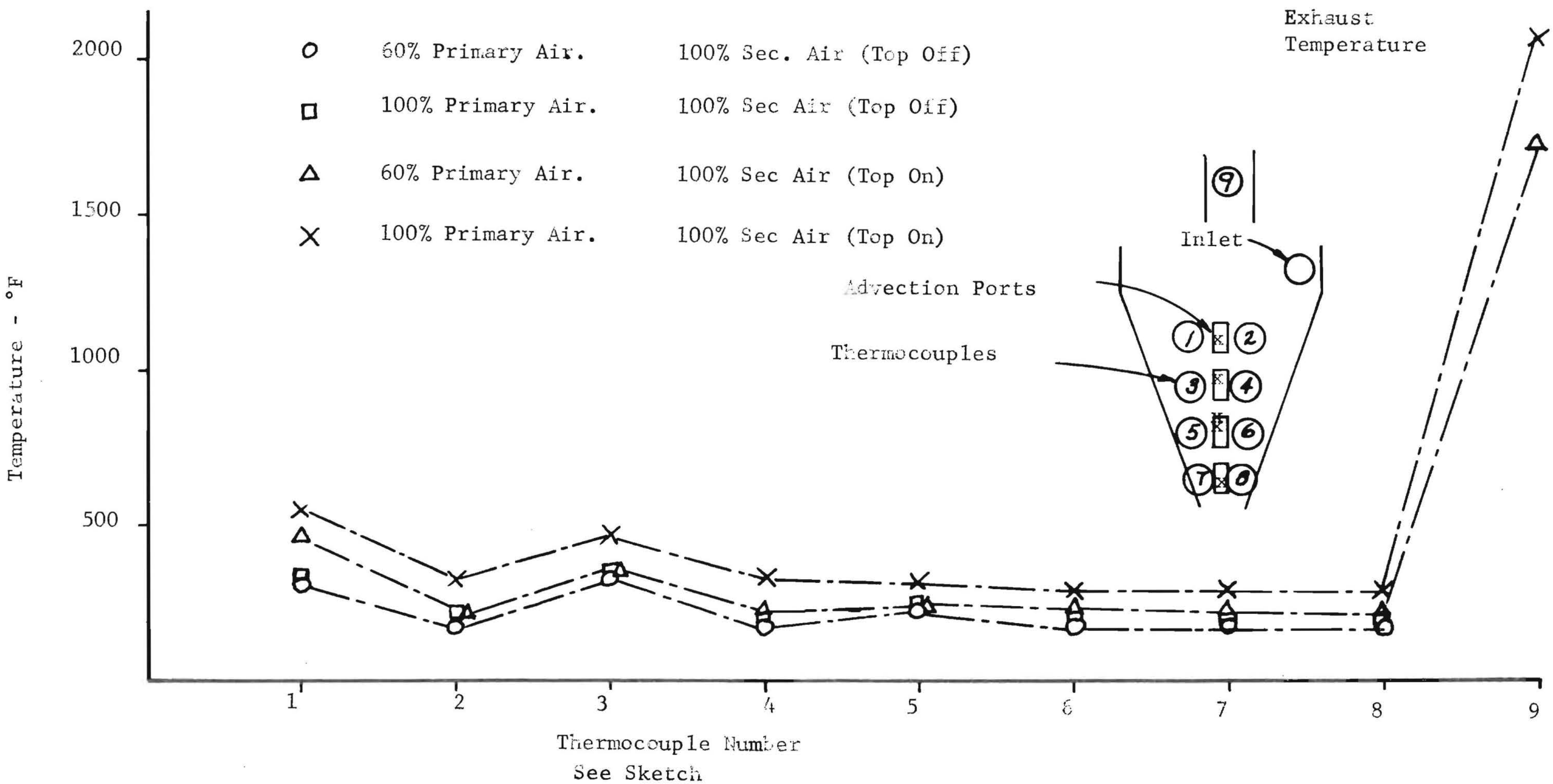


Figure 8. Temperature Distribution Across Advection Ports - Forced Secondary Air

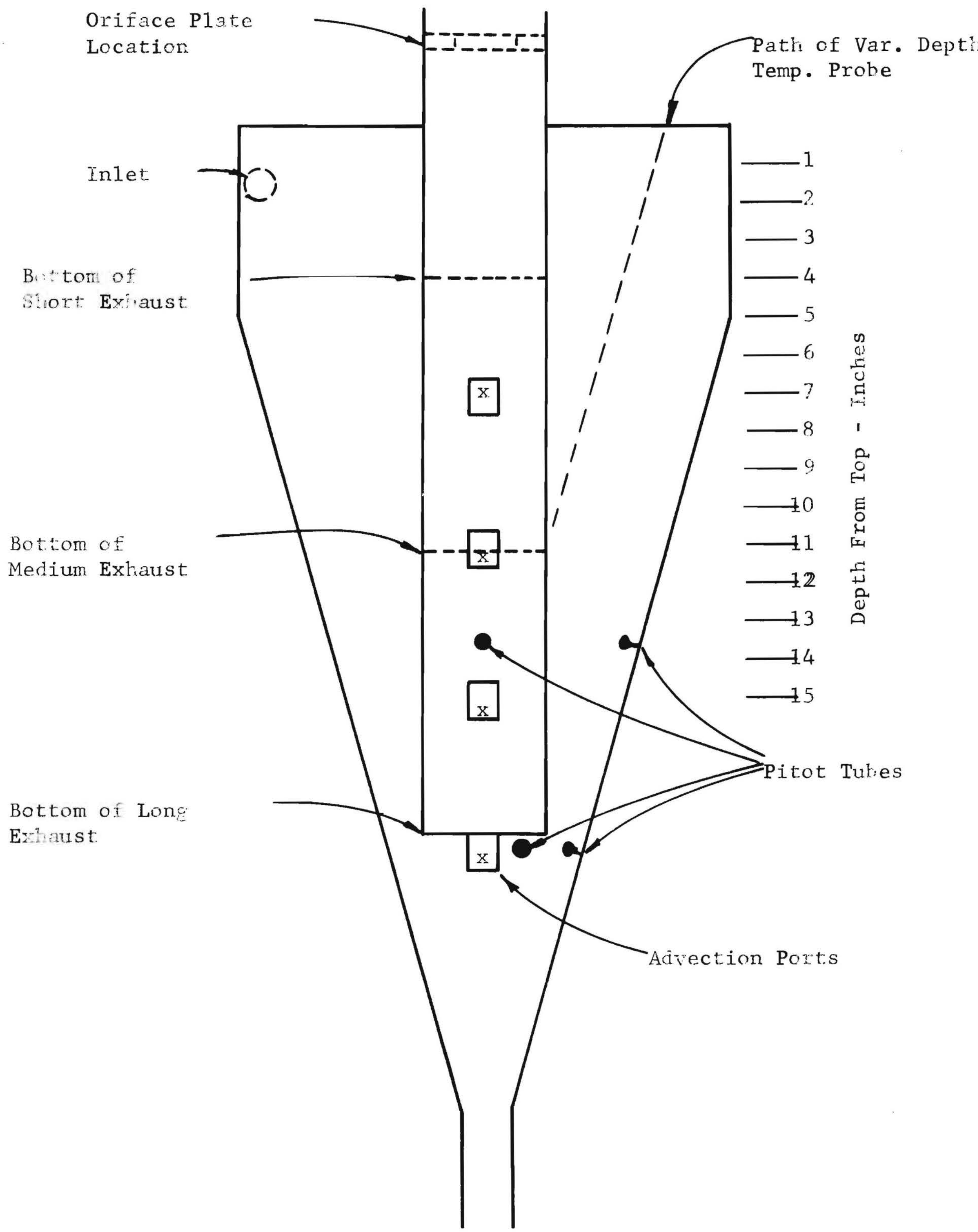


Figure 9. Burner with Variable Depth Temperature Probe

It appeared at this point that the secondary air introduced along the burner wall not only was very effective in keeping the wall cool, it would also keep the peanut hulls too cool to burn properly. Further, introduction of the auxiliary air in this manner possibly could have contributed to the short circuiting of the hot combustion gases. The outer can was removed from the burner and thin stainless steel patches were spot-welded over each of the advection ports. Care was taken to ensure that the patches sealed the ports. The plates welded over the ports may be seen in Figure 7.

Tests of the burner with this modification could not be accomplished because it was found impossible to ignite the primary air-fuel mixture in the chamber. Adjustment of mixing valve had no effect on the condition. It appeared that additional air was necessary for combustion in the chamber. A second inlet port similar to the primary air inlet was added. This air inlet was later used to introduce peanut hulls and will be referred to as the peanut air inlet. This inlet was placed 90° from the primary air inlet. The additional air from this inlet corrected the ignition problem.

Figure 10(a) shows the temperature as a function of slant distance as measured by the top variable temperature probe. Two primary air flow rates are shown. No auxiliary air was used in these tests. Although the temperature profile is relatively flat, the maximum temperature is reached at a depth near the lower end of the exhaust stack indicating that there is still the tendency towards short circuiting. The stack was modified and tested at two other lengths in an attempt to force the vortex lower in the burner. Figure 9 shows the exhaust lengths for all three cases. Figures 10(b) and 10(c) show the temperature profile for the medium and long length exhaust. These curves are for two primary air flow rates with the peanut air at 47 feet/second. Figure 11 shows

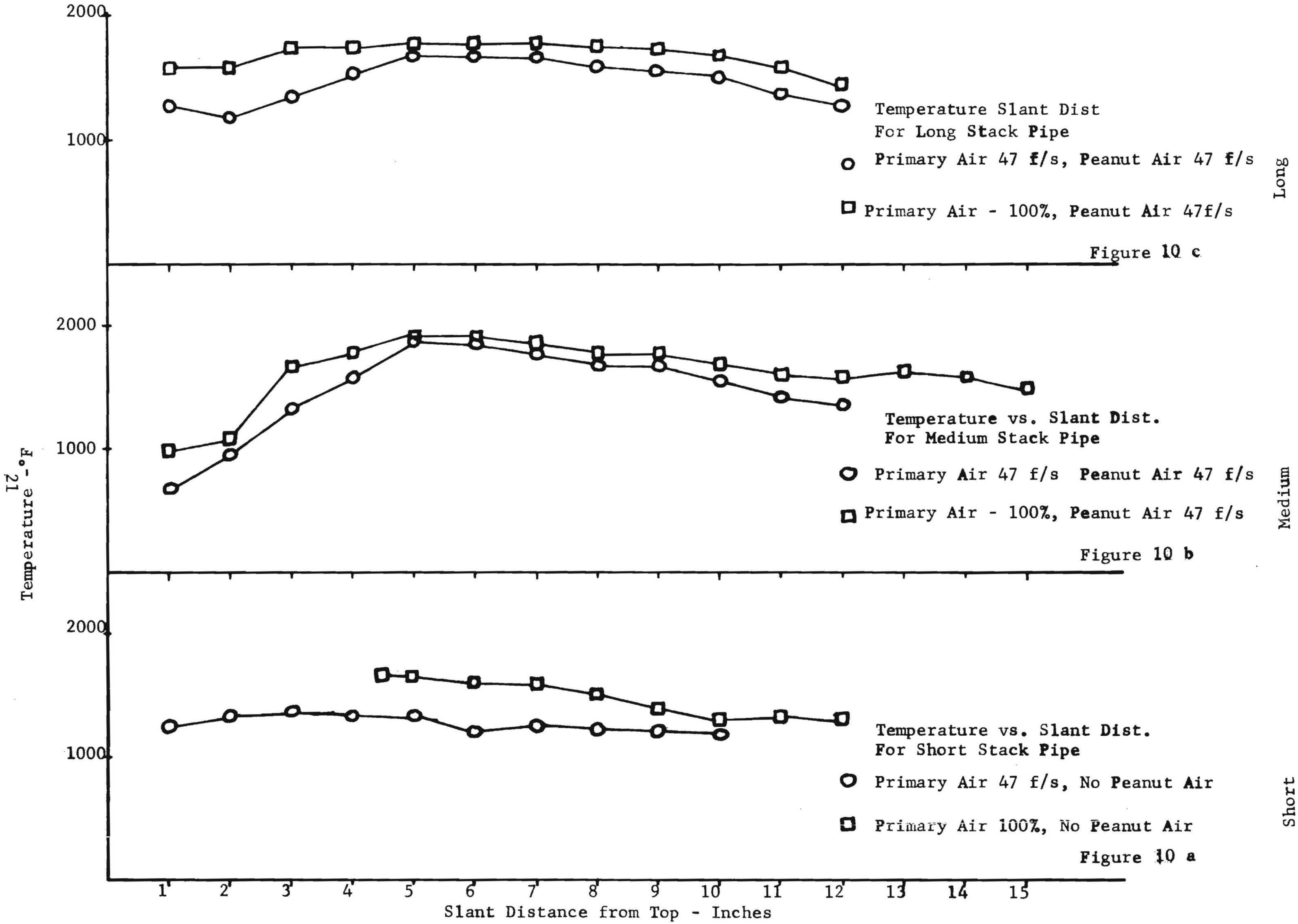
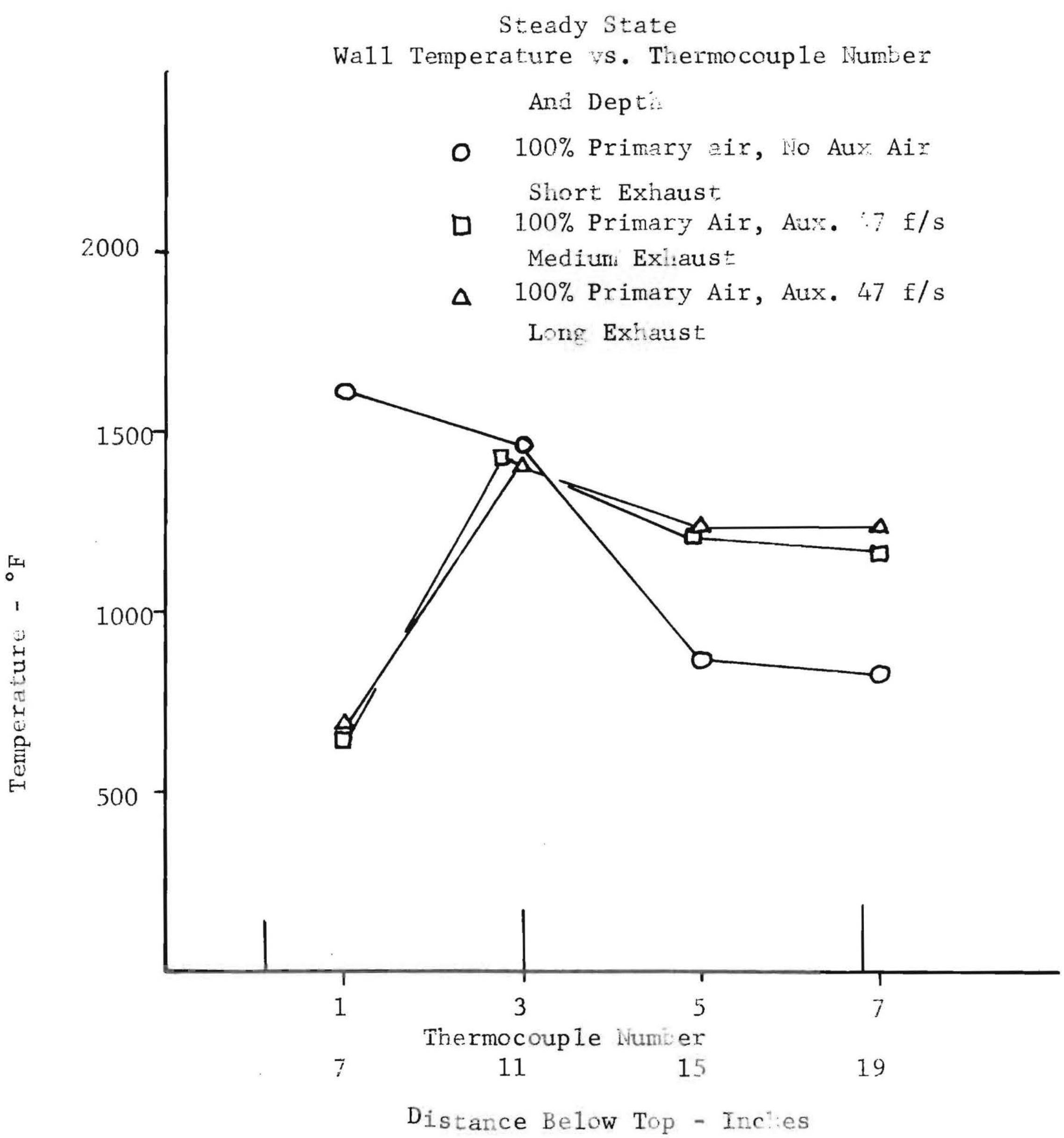
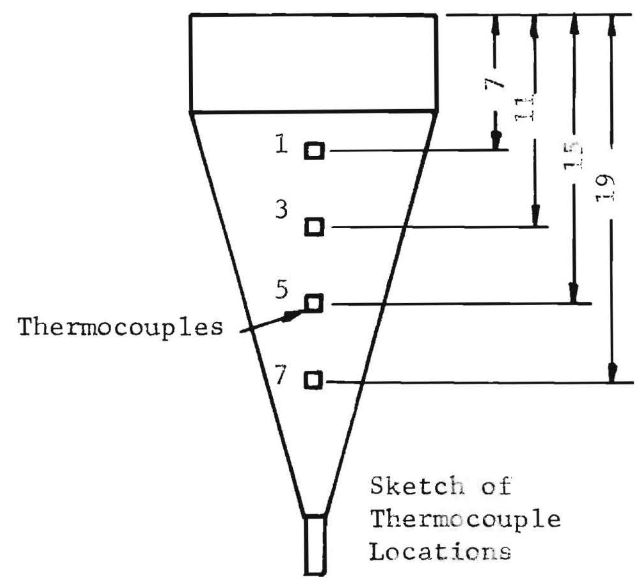

Temperature -°F
2000
1000
Temperature Slant Dist
For Long Stack Pipe
Primary Air 47 f/s, Peanut Air 47 f/s
Primary Air - 100%, Peanut Air 47f/s
Figure 10 c
Long
Temperature vs. Slant Dist.
For Medium Stack Pipe
Primary Air 47 f/s Peanut Air 47 f/s
Primary Air - 100%, Peanut Air 47 f/s
Figure 10 b
Medium
Temperature vs. Slant Dist.
For Short Stack Pipe
Primary Air 47 f/s, No Peanut Air
Primary Air 100%, No Peanut Air
Figure 10 a
Short
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
Slant Distance from Top - Inches

Figure 10



Steady State
Wall Temperature vs. Thermocouple Number
And Depth
100% Primary air, No Aux Air
Short Exhaust
100% Primary Air, Aux. 47 f/s
Medium Exhaust
100% Primary Air, Aux. 47 f/s
Long Exhaust
2000
1500
1000
500
Temperature - F°
1
3
5
7
Thermocouple Number
7
11
15
19
Distance Below Top - Inches



7
11
15
19
1
3
5
7
Thermocouples
Sketch of
Thermocouple
Locations

Figure 11

the effect of stack length on chamber wall temperatures. Some increase in temperatures in the lower regions was noted, but very little.

A series of tests were conducted for medium exhaust length with the exhaust outlet area restricted to 50% and 25% of the original area. This restriction was accomplished with orifice plates as shown in Figure 9. The purpose of these restrictions was to restrict the flow and reduce the tendency toward short circuitry.

Figures 12(a) and 12(b) show the results from these tests. At first there appears to be little measurable difference due to the restriction. However, careful examination reveals there is a minor reduction in short circuiting for high primary air flow rates. Figure 13 shows the effect of restriction on the burner wall temperatures.

D. Peanut hull tests

Although the model was never intended to burn peanuts, several tests were conducted with peanuts to determine whether meaningful data could be obtained. The peanut hulls were crushed to increase the surface area to volume ratio. This was considered necessary in the model due to the much shorter residence time.

The second primary inlet air port was used to introduce the peanut hulls. Figure 14 shows this design and the means used to introduce peanut hulls. This air is referred to as peanut air henceforth. For the initial tests with this design no peanuts were placed in the hopper so that only auxiliary air was introduced through this port.

The primary inlet was set at approximately 85 feet/second and the incinerator allowed to reach a steady state, i.e., all temperatures were allowed to stabilize. The auxiliary air blower was turned on and a mixture of peanut hulls

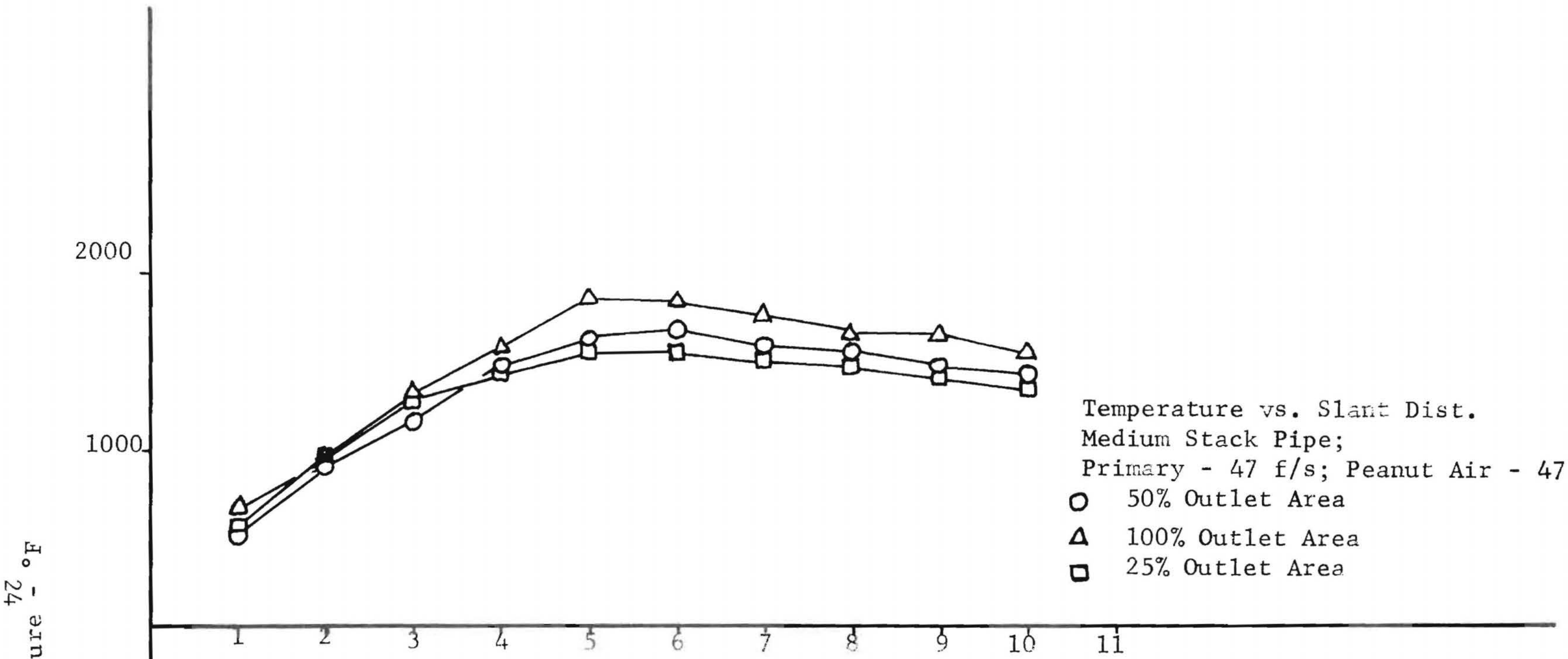


Figure 12 b

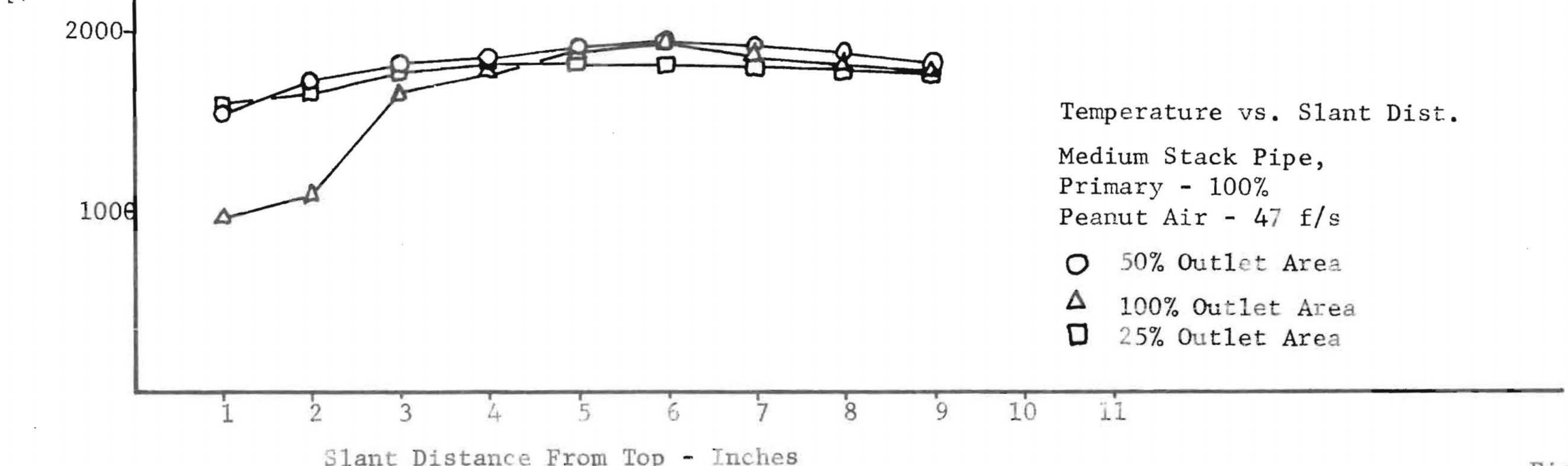


Figure 12 a

Wall Temperature vs. Thermocouple Number
And Depth
Medium Exhaust, 100% Primary Air, 47 f/s Aux. Air.

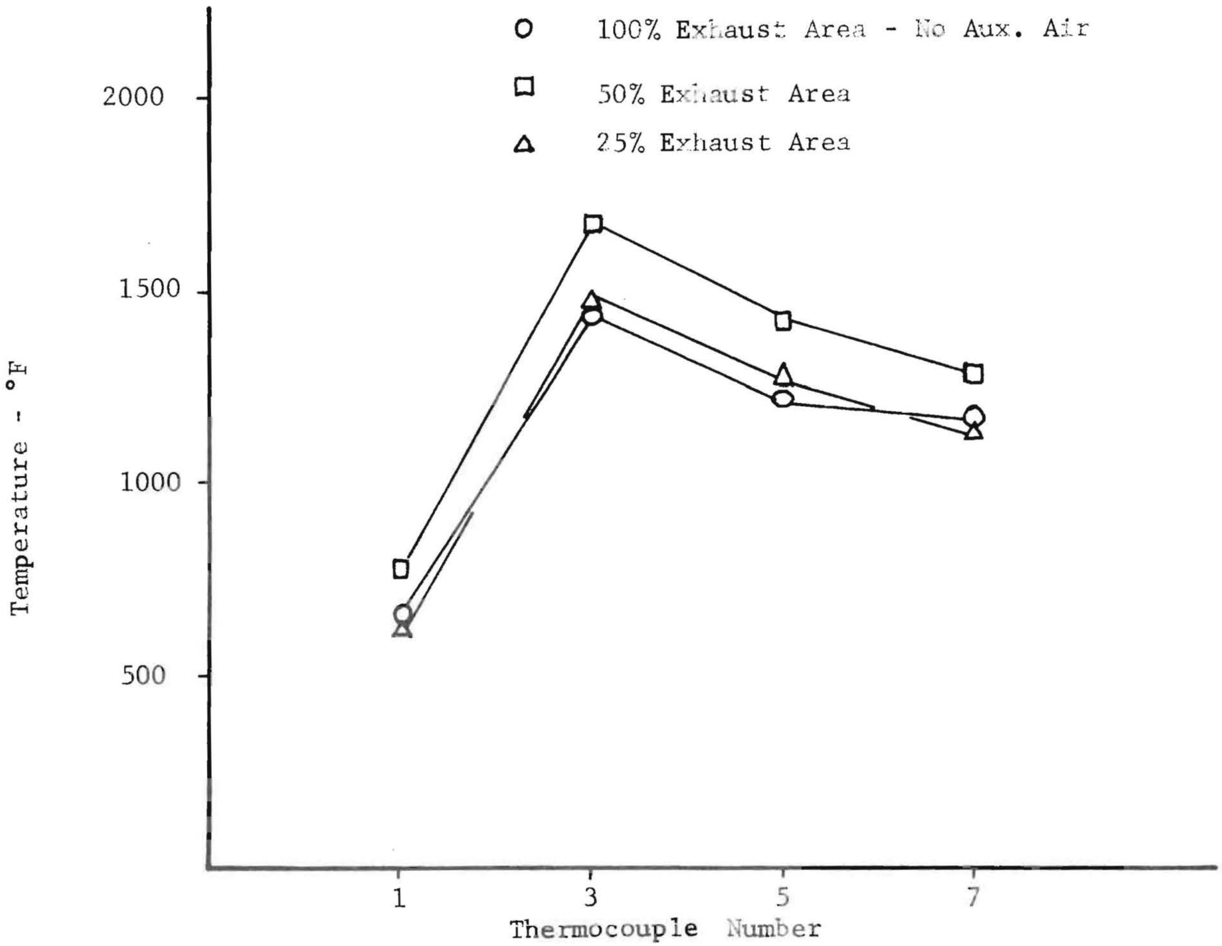


Figure 13

and air were blown into the chamber at a velocity slightly greater than 85 feet/second. A temperature rise from 1000° F to 1300° F was observed when the peanut hulls were introduced. Once the temperatures had restabilized the primary air was gradually turned off. Temperatures decreased and burning ceased as the primary air was stopped. The test was repeated using several different techniques* in an attempt to get an even flow of the peanut hulls in order to maintain combustion. All attempts were unsuccessful in maintaining peanut hull combustion.

Since the primary air-fuel mixture had been purposely run rich, it was decided to repeat the above tests with the exception that only air would be introduced through the peanut port. The temperature rise from 1000° F to 1300° F was again observed indicating that more complete combustion of the primary air-fuel mixture was the source of the temperature rise rather than combustion of the peanut hulls. Further verification was obtained when the contents of the solid particle discharge port was examined. The peanut hulls separated from the exhaust gases by the vortex action in the burner were partially charred. Only a fraction of the hulls had been burned, which is not surprising considering the short contact (residence) time.

E. Cylindrical tube with plug

Since a stable vortex had not been established and maintained with the 15° half-angle cone shape, a different approach was considered. A transparent model with the use of smoke injection and flow direction indicators appeared to be necessary to study vortex formation and stability.

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*Funnel-type feeding.

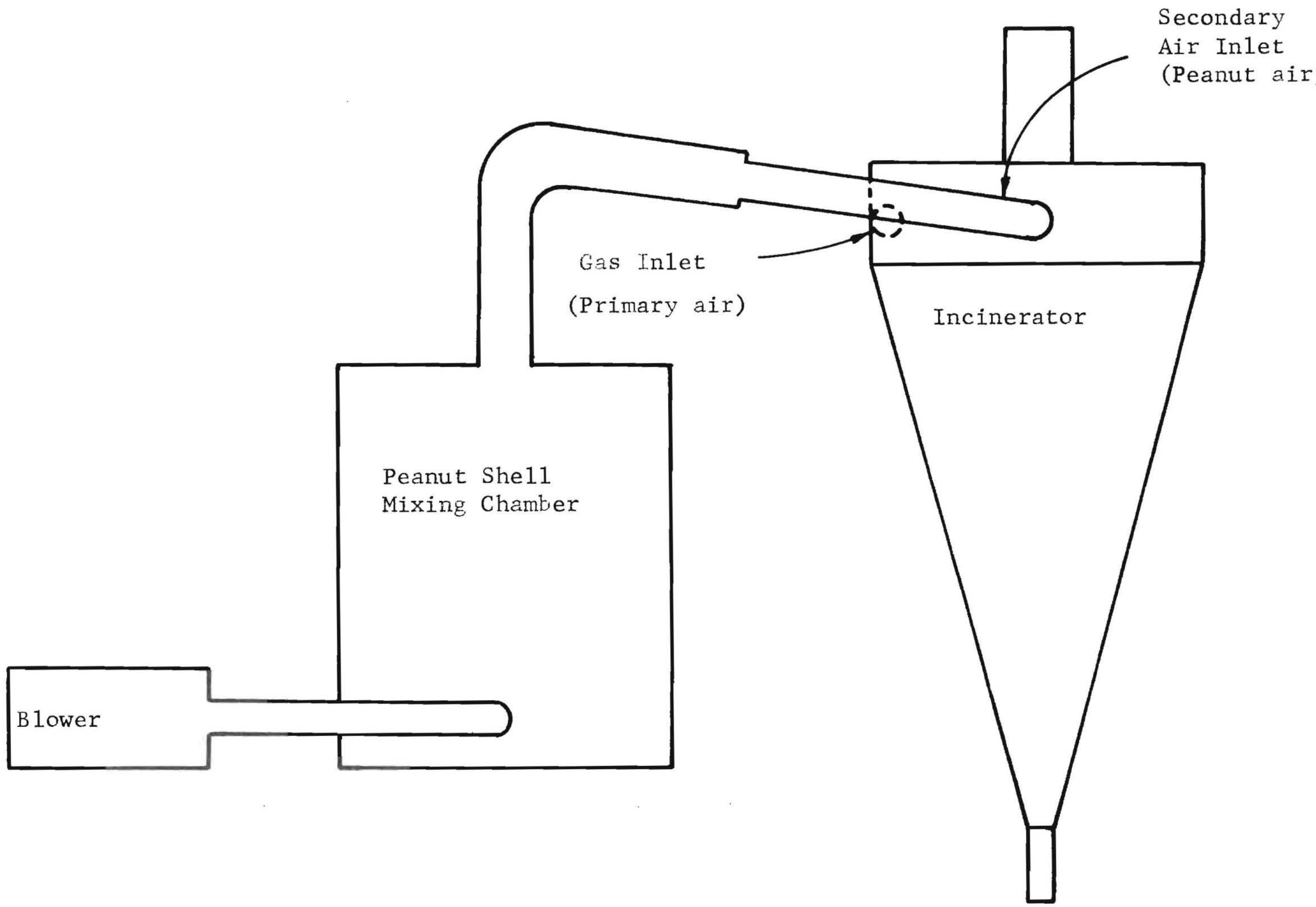

Figure 14. Conical Model Adapted for Burning Crushed Peanut Hulls.

The cylindrical plexiglass tube (8 in. diameter) appeared to be the most convenient and readily adapted, using a geometrical bottom plug to turn the flow. This model was prepared with three bottom plug designs turned from wood.

The tests with the tangentially-fed circular cylinder ($L/D \approx 5$) using a bottom plug to turn the flow indicated that vortex formation was readily obtainable, an outer rotational flow on the innter wall of the cylinder was always present. With a particular design on the bottom plug, a very high speed inner vortex could be obtained which was attached to the tip of the plug and extended upward to the exhaust opening.

The model used is shown in Figure 15. Flow visualization was accomplished* by smoke injection, flow direction indicators and a rotating string probe with a stroboscope. Direct velocity measurements in swirl-type flow are extremely difficult to accomplish with accuracy. They were not attempted in these tests. The rotational velocities at the outer edge determined with a rotating string and stroboscope are considered satisfactory for the preliminary work. It is noted that the ratio of vortex length to maximum diameter was 14 in these tests with the cylindrical tube.

One interesting feature of the bottom plug is that in the long radius (dished) turning of the flow, a turbulent "mixing" or "turnover" zone is formed where particles of reasonable size are kept in constant motion.

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*Mr. M. D. Bowen supervised these studies.

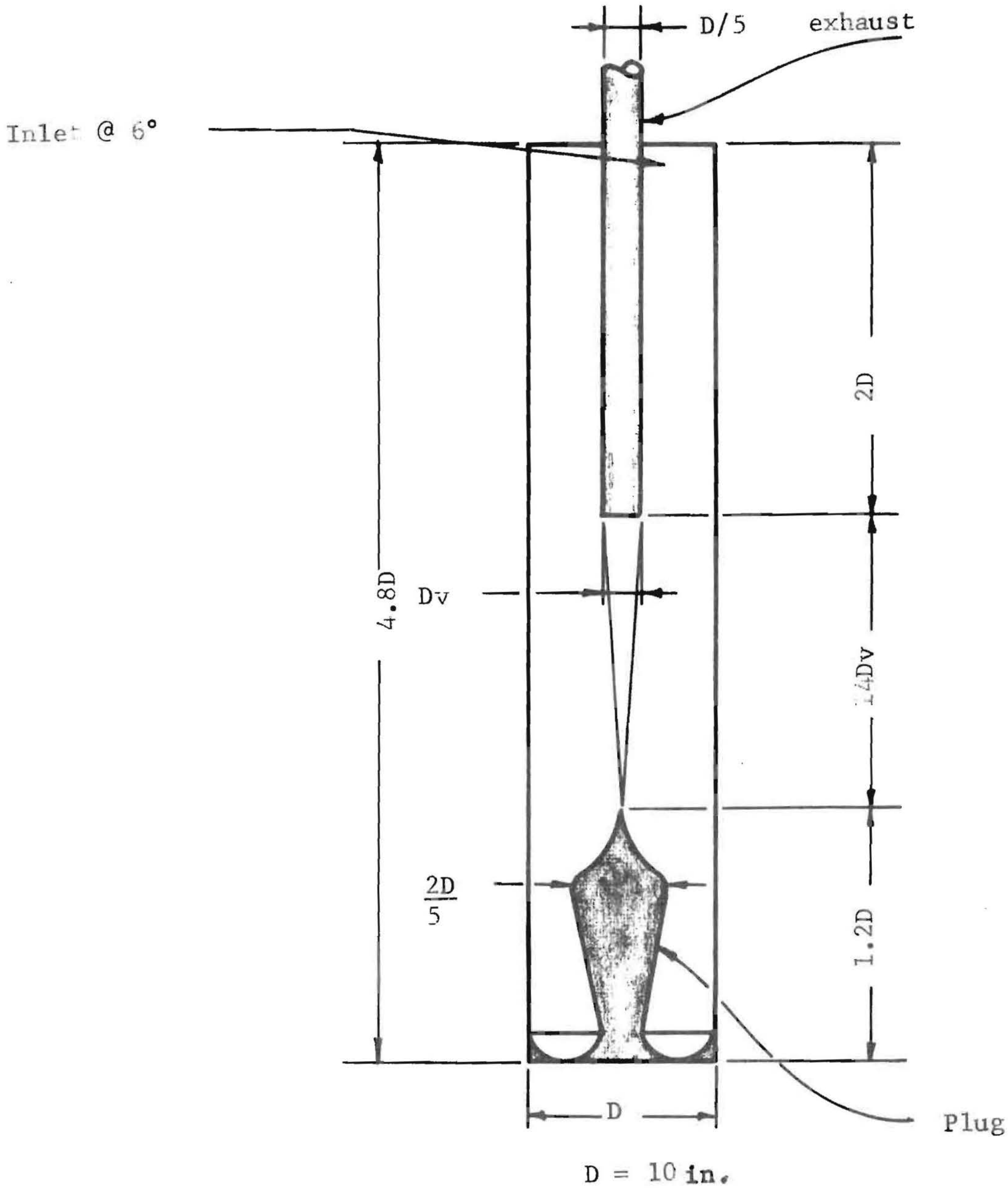

Figure 15. Flow Visualization Model

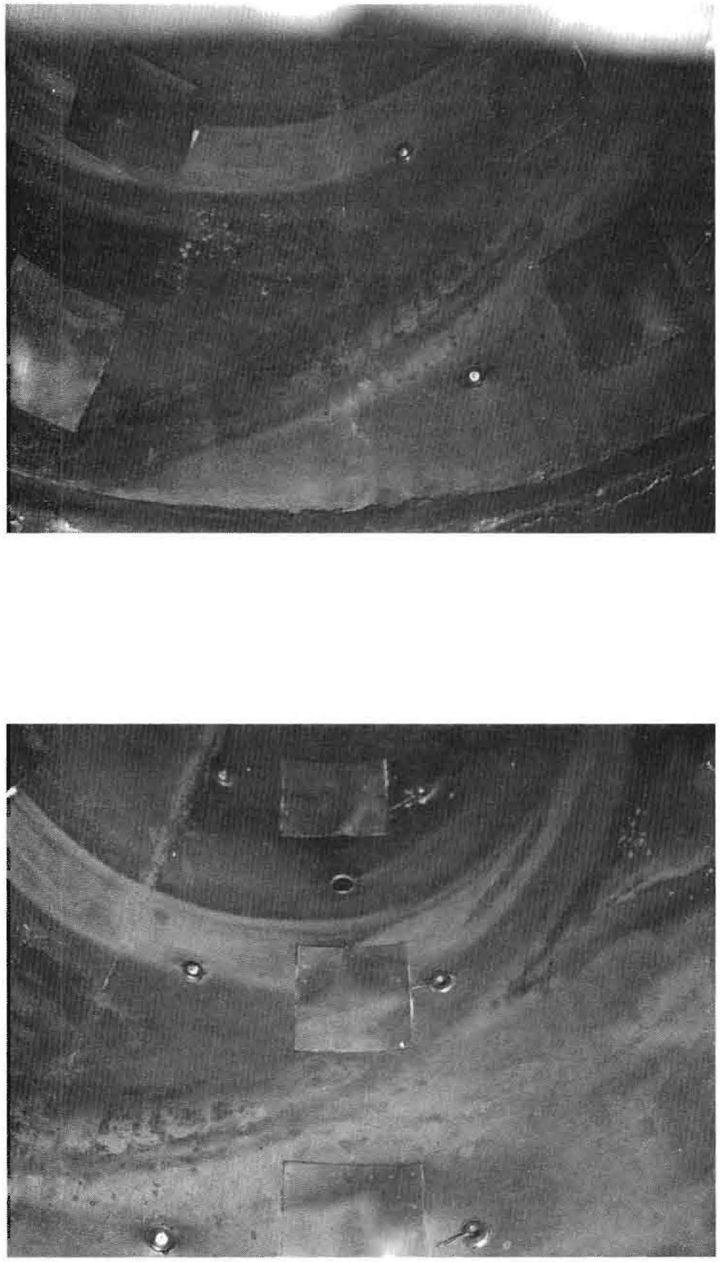

Figure 16. Pictures of the Inner wall of the Conical Model.

a) A small circular primary air inlet. The initial work had a 3/4 inch pipe tangentially feeding air. Later, this same pipe fed the gas-air mixing valve and the entrance into the 'burner' was somewhat larger. However, most air cyclone equipment uses a rectangular, tangential inlet with a height approximately one-half the cyclone major diameter, and a width approximately one-fourth the cyclone major diameter.

b) The length of the straight section was approximately one-half the cyclone major diameter - which is generally of the order of twice the major diameter. The longer length of the straight section would enable time for some stabilization of the flow before it enters the cone.

c) The cone angle used was 30° total angle. This may be too large. Again, many practical cyclones use angles of only about 20°. The larger angle could have either produced a high inward acceleration of the flow or created flow turbulence, both reasons for the tendency for short-circuiting of the flow out the exhaust rather than traveling the full route through to the bottom of the cone.

The data presented in Figures 10 and 11 suggest that at least with <u>no</u> secondary air, and with the higher primary air flows, a reasonable temperature distribution was obtained, at least in these tests. Apparently, secondary air added in the lower parts of the cone was unable to be used by the fuel-rich gases to increase combustion efficiency. However, no attempt was made to use only the top tier of advection ports for secondary air. This might have improved the efficiency of burning the fuel-rich primary mixture in a manner similar to the use of peanut air to improve the efficiency of burning.

If the primary gas-air mixture entered at approximately 85 ft/sec, and made say three revolutions in the model this would correspond to roughly

$$\frac{3 \times \pi \times 1.04}{85} = 0.115 \text{ sec residence time.}$$

Thus, the lack of burning of the peanut hulls is not surprising. Actually, if burning had been achieved, it would have been surprising. However, the primary purpose of trying the peanut hulls was to check on the separation effectiveness - since the viscosity of gases increases with temperature.

Appendix C briefly presents some additional information regarding peanut hulls developed at Georgia Tech. 'Coking' of peanut hulls is essentially exothermic once the hull temperature is up to 500-700°F. However, burning the 'coke' to ash will require a higher temperature and additional air to complete the ashing operation in any reasonable time, and will probably require residence times of one minute or more. Consequently, a turbulent "turn-over compartment" would seem to be advisable. High velocities to obtain separation are not conducive to long residence times in equipment of reasonable size.

The plastic cylindrical flow model appeared to have a fairly even distribution along the walls. This could well have resulted from the fact that the L/D ratio of the straight walls was large enough to permit the flow from the 3/4 inch diameter air inlet to redistribute itself unevenly over the walls.

The contoured bottom plug permitted smooth turning of the flow into the inner high velocity tight vortex, but also provided a turbulent "turnover compartment" in the dished section of the plug. The phenomena associated with the plug in turning the flow is not completely understood at present. However, three plugs were made and all worked - some better than others admitedly It would appear that the plug could be readily used also in the conical cyclone, perhaps with dramatic results. However, to optimize the design, a flow model should be made to study to application of the plug to the condition where the flow approaching the plug is conical.

VI. CONCLUSIONS

It is concluded that:

1. Natural advection of secondary air is not practical in the model tested and probably not in any model because of the high centrifugal pressure.

2. Forced advection of secondary air in the model tends to encourage short-circuiting of the cyclonic flow out the stack in the model tested.

3. The higher temperature creating less dense gases and a higher gas viscosity does not appear to be detrimental - at least in the range to 1300° F.

4. Peanut hulls are readily separated in hot gases obtained from burning Atlanta City gas with air.

5. Residence times greater than 1/10 second are required to char peanuts hulls in the model (1 ft. diameter) cyclone 'burner' with 1000° F gases. Burning to ash will require longer residence time and probably higher temperature.

6. A cylindrical cyclone with contoured bottom plug may be better adapted for this type service.

7. A turbulent "turnover compartment" may be the compromise to burning materials such as peanut hulls which require a significant residence time to ash and still maintain velocities to insure particulate removal from the exhaust gases.

VII. RECOMMENDATIONS

It is recommended that:

1. The next phase should include two model "burners", a cylindrical burner with contoured plug and a conical burner with more conventional proportions and a contoured plug.

2. The burners be constructed without secondary air ports initially.

3. Work with a large 'burner' be delayed until the preliminary work in 1 (above) is completed to determine which type burner should be constructed.

APPENDIX A

PEANUT HULL INCINERATOR STUDY
C. W. Bouchillon, P.E.

I. Introduction

The proposed design configuration is for an incinerator to be used in burning peanut hulls in such a manner that there will be no significant air pollution resulting from its operation. The following design parameters were established in conference with Mr. Jack Kinney, Mr. John Sims and Mr. Max Akridge on April 9, 1968.

1. Quantity of Material to be Burned - 2000 - 8000 lbm/hr.
2. Physical characteristics of the material
   a. Initial particle size - 50μ - 3/8 x 3/8 - 3/8 x 1.0
   b. Surface area/weight - 10 in^2/gm
   c. Specific gravity - 0.2 - 0.25
   d. Heating value - 9000 Btu/lbm
   e. Ash content - 2.5%
   f. Softening temperature of ash - 2100° F
   g. Fluid temperature of residue - 2150° F
3. Required separation effectiveness - 85 - 95%
4. Energy for forced ventillation - economically justifiable
5. Discharge - to atmosphere
6. Air to fuel ratio - 16 to 1 (160% excess air)

II. Flow rate predictions

The required air flow rates under the conditions enumerated above are

INLET AIR

$$\dot{M}_A = 2000 \text{ lbm/hr} \times 16 \frac{\text{lbmA}}{\text{lbmF}} = 32{,}000 \frac{\text{lbmA}}{\text{hr}} \quad (1)$$

$$\dot{M}_{AMAX} = 8000 \frac{\text{lbmF}}{1 \text{ hr}} \times 16 \frac{\text{lbmA}}{\text{lbmF}} = 128{,}000 \text{ lbmA/hr.} \quad (2)$$

DISCHARGE GASES

$$\dot{M}_{FG} = 2000 \frac{\text{lbmF}}{\text{hr}} \times 17 \frac{\text{lbmFG}}{\text{lbmF}} = 34{,}000 \frac{\text{lbmFG}}{\text{hr}} \quad (3)$$

$$\dot{M}_{FGMAX} = 8000 \frac{\text{lbmF}}{\text{hr}} \times 17 \frac{\text{lbmFG}}{\text{lbmF}} = 136{,}000 \frac{\text{lbmFG}}{\text{hr}} \quad (4)$$

An approximate analysis of the combustion process for the discharge temperature conditions (neglecting pressure changes) may be made as follows.

$$\dot{M}_{A_1} h_{A_1} + \dot{M}_{F_1} h_{F_1} + \dot{M}_F QF_1 = \dot{M}_{FG_2} h_{FG_2} \quad (5)$$

Assuming that

$$h_{A_1} \simeq C_p T_1 = 0.24(530) = 127 \text{ Btu/lbm}$$

$$h_{F_1} \simeq 0$$

$$Q_F = 9000 \text{ Btu/lbm}$$

$$\dot{M}_{A_1} = 32{,}000 \text{ lbmA/hr}$$

$$\dot{M}_{F_1} = 2000 \text{ lbmF/hr}$$

$$h_{FG_2} \simeq C_p T_2 = \frac{1}{17} [16(127) + 9000] = 650 \quad (6)$$

$$T_2 \simeq \frac{650}{0.24} = 2700° \text{ R} = 2240° \text{ F} \quad (7)$$

The inlet pressure should be limited to a reasonable level and possibly such that only primary starting air and conveying air be required to be pumped. Consider the configuration of the gasoline blow-torch. Conservation of momentum suggests that the base end be of solid construction but that the entrance ports for the primary air supply be such that a tangential motion be imparted to the flow as it enters the burning region. A sketch is presented as Figure 1 to illustrate this principle.

Let us assume that the inlet stream of air/peanut hulls may be satisfactorily conveyed at a loading rate of 0.5/lbm of hulls per lbm of air. The advection ports then must supply the additional 14 pounds of air per lbm of hulls to support the combustion process. The ports should be shaped such that they should be shaped such that they are in the streamline of the main-flow of air. In order to establish this direction, let us assume that the velocity in a vortex with a constant density fluid may be approximated by

$$u \simeq -C_1 \frac{r}{z} - C_2 \qquad (8)$$

where C_1 is a constant for a given total throughflow and C_2 is the equivalent radial inflow velocity due to the advection ports and assumed uniform over the wall area. This necessitates varying their sizes which will be considered later.

Assume that the tangential velocity is given by

$$v = \frac{C_3}{r} \qquad (9)$$

which approximates the free vortex flow characteristic of cyclones near the wall.

Assume that the vertical velocity is given approximately by

$$w = C_1 \left[2 \ln \frac{z \tan B}{r} - 1\right] + C_2 \frac{z}{r} \qquad (10)$$

which satisfies the continuity criterian for constant density fluids in cylindrical coordinates.

The volumetric rate of flow will have to increase by a factor of $\frac{2700}{530}$ or 5.10 times the volumetric inlet flow rate.

The first question which arises is to determine a discharge volumetric flow rate which will allow reasonable pressure levels to exist within the combustion chamber. Assuming for the moment that atmospheric pressure prevails, then the discharge area must be such that subsonic flow prevails. If the dynamic head is limited to 1.0 psi, then the equivalent air velocity would be approximately

$$1/2\ \rho\ \frac{V_2^2}{g^o} + (P_2 - P_1) = 0 \tag{11}$$

$$V_2^2 = 2 \times 1\ \frac{\text{lbf}}{\text{in}^2}\ \hat{v}\ \frac{\text{ft}^3}{\text{lbm}} \times 32.2\ \frac{\text{lbm ft}}{\text{lbm sec}^2} \times 144\ \frac{\text{in}^2}{\text{ft}^2}$$

$$\hat{v} = \frac{53.3\ (2700)}{15.7 \times 144} = 63.8\ \frac{\text{ft}^3}{\text{lbm}}$$

$$V_2^2 = 59 \times 10^4\ \frac{\text{ft}^2}{\text{sec}^2}$$

$$V_2 \simeq 770\ \frac{\text{ft}}{\text{sec}} \tag{12}$$

The discharge area

$$\dot{M} = \frac{AV_2}{\hat{v}}\ ,\ \text{or}\ A_2 = \frac{\dot{M}\hat{v}}{V_2}$$

$$A_2 = \frac{34{,}000\ \frac{\text{lbm}}{\text{hr}} \times 63.8\ \frac{\text{ft}^3}{\text{lbm}}}{770\ \text{ft/sec} \times 3600\ \frac{\text{sec}}{\text{hr}}} = 0.785\ \text{ft}^2$$

$$A_2 = 0.785\ \text{ft}^2 \tag{13}$$

Double the area and reduce the flow velocity by 2.0 or say make the discharge area 2.0 ft^2. The discharge velocity would then be an average of 304 ft/sec. The equivalent dynamic head would then be reduced to $\triangle P$ = 0.157 psi which is compatible with blower capabilities. This is equivalent to a radius of

$$A = \pi R_o^2$$

$$R_o = \sqrt{\frac{2.0}{3.14}} = (0.636)^{1/2} = 0.8 \text{ ft} \tag{14}$$

$$D_{outlet} = 1.6 \text{ feet}$$

$$D_{burner} = 4.0\ D_{outlet} = 6.4 \text{ feet} \tag{15}$$

$$\text{Length of Cyclone} = 12 \text{ feet} \tag{16}$$

$$L = \frac{D_b}{2} \frac{1}{\tan 15°} \tag{17}$$

The equilibrium surface of particles may be obtained by setting the drag force on the particle equal to the centrifugal force on the particle and ascertain its equilibrium radius.

The drag force is given by

$$F_d = C_D\ \pi a^2 \rho\ \frac{u^2}{2} \tag{18}$$

for a sphere and the centrifugal force is given by

$$F_c = 4/3\ \frac{\pi a^3}{r}\ (\rho_s\ V_s^2 - \rho v^2) \tag{18A}$$

Observing that for the equilibrium condition i.e. when $F_d = F_c$, the particle is stationary in r, and $v^2 \simeq V_s^2$, equating the two forces yields

$$C_D \frac{\pi a^2 \rho u^2}{2g_o} = 4/3\ \pi\ a^3 \frac{v^2}{r} \frac{\rho_s - \rho}{g_o} \tag{19}$$

A reasonable approximation to the drag coefficient may be obtained by

$$C_D = \frac{24.0}{R_e} \text{ for } 0 < R_e < 1.0 \tag{20}$$

and

$$C_D = \frac{19.5}{R_e^{4/7}} \text{ for } 1.0 \le R_e < 1000 \tag{21}$$

where

$$R_e = \frac{2\ \rho u\ a}{\mu} \tag{22}$$

These relations may be combined with equation 19 to determine the operating characteristics of the cyclone.

In order to account for the variable density due to burning, assume that

$$\rho \simeq C_4 r \tag{23}$$

Then modified velocity profiles may be postulated as

$$u = -\frac{C_1^1}{z} - \frac{C_2^1}{r} \qquad\qquad v = \frac{C_3^1}{r^2} \tag{24}$$

and

$$w = C_1^1 \left[\frac{3}{r} \ln\left(\frac{z\tan\beta}{r}\right) - \frac{1}{r}\right] + \frac{C_2^1\ z}{r^2} \tag{25}$$

This reflects a different flow configuration but one in which as the centerline is approached, the vertical velocities are much higher as would be anticipated. These profiles may be used in obtaining estimates of the equilibrium radius.

The angle of the inlet louvers should be such that

$$\alpha = \text{ARCTAN}\ \frac{w}{-v} \tag{26}$$

or

$$\alpha = \text{ARCTAN} \left\{ \frac{C_1^1 \left[\frac{3}{r} \ln \left(\frac{z\ ta\beta}{r} \right) - \frac{1}{r} \right] + \frac{C_2^1\, z}{r^2}}{-(C_3^1/r^2)} \right\} \tag{27}$$

Also with $r = z \tan \beta$,

$$\alpha = \text{ARCTAN} \left[\frac{+C_1^1\ r - C_2^1\ z}{C_3^1} \right] z$$

or

$$\alpha = \text{ARCTAN} \left[\frac{+C_1^1 \tan \beta - C_2^1}{C_3^1} \right] z \tag{29}$$

The angle of the louver would thus decrease with the decreasing values of z. Values for C_1^1, C_2^1, and C_3^1 will be determined later.

Louver sizing is a very complex problem and should be approached with rough estimates at the outset. The assumed radial velocity distribution of

$$u = -\frac{C_1^1}{z} + \frac{C_2^1}{r} \tag{30}$$

evaluated at the location

$$r = z \tan \beta$$

$$u = -\frac{C_1^1}{z} + \frac{C_2^1}{z \tan\beta} \tag{31}$$

Therefore for small z, it is anticipated that more flow is anticipated. This will be true because of the advection effects of the increased velocity due to the free vortex phenomena.

Therefore, louvers of the same area may be used at the outset as a reasonable design criterion.

The sizing of these louvers is indeed a troublesome problem. However, to give something to shoot for, the inlet velocity may be assumed to be twice the equivalent unburned exit velocity, or 0.4 times the exit velocity of 304 ft/sec.

This would be 120 ft/sec or a dynamic pressure requirement of 0.124 psi or 3.4 inches of water which is quite reasonable for a blower requirement.

The primary air inlet size should be

$$A = \frac{\dot{m}\,\hat{v}}{V_i} = \frac{2000 \text{ lbm} \times 12.6 \text{ ft}^3}{120 \frac{\text{ft}}{\text{sec}} \text{ hr } 3600 \frac{\text{sec}}{\text{hr}} \text{ lbm}} = 5.82 \times 10^{-2} \text{ ft}$$

$$A = 0.0582 \text{ ft}^2 = 8.35 \text{ in}^2 \tag{32}$$

For a round pipe,

$$\frac{\pi D^2}{4} = 8.35 \text{ in}^2$$

$$D^2 = \frac{8.35 \times 4}{\pi} = 10.6$$

$$D_i = \underline{3.25 \text{ in}} \tag{33}$$

D = 3.25 inches - For a circular hole

Try 2 lbm of air in conveying line for each lbm of hulls

then

$$A = 16.7 \text{ in}^2$$

$$D^2 = \frac{16.7 \times 4}{\pi} = 21.2$$

$$D_i = 4.60 \text{ inches} \tag{34}$$

At the outset,

<u>Try $D_i = 4.0''$ for conveyor for feeding hulls</u>

Then if because of the burning effect reduces the density such that

$$v = \frac{C_3^1}{r^2}$$

or

$$C_3^1 = v_i r_i^{\ 2} = 120 \text{ ft/sec} \times (3.2)^2 \text{ ft}^2 = 1230 \frac{\text{ft}^3}{\text{sec}}$$

So that

$$v = \frac{1230}{r^2} \tag{35}$$

However, the wall losses would probably reduce this to a tangential velocity such that

$$v \simeq \frac{800}{r^2} \tag{36}$$

Assuming that there are louvers every two feet of length for ten feet and spaced on 90° spacings, there would be 20 louvers at five locations. This would mean that for any radius, the mass input may be assumed to be

$$\dot{m} = \frac{A_L V}{\hat{v}}$$

The advected air may be determined from

$$\Sigma \dot{m}_i = 28000 \frac{\text{lbmA}}{\text{Hr}}$$

Remembering that

$$v \simeq \frac{800}{r^2} = \frac{800}{z^2(\tan^2 15°)} = \frac{11{,}100}{z_i^2} \tag{37}$$

then for the five levels of four louvers,

$$\Sigma \dot{m}_i = \sum_{i=1}^{5} \frac{4A_L v}{\hat{v}} = \frac{4A_L\ 11{,}100}{(12.6) \times 3.2} \sum_{i=1}^{5} \frac{R_i}{z_i^2} = 28{,}000 \tag{38}$$

$$A_L = \frac{28{,}000(12.6)(3.2)}{3600 \times 11{,}100 \times 4} \cdot \frac{1}{\sum_{i=1}^{5} \frac{1}{z_i^2}}$$

taking

$$z_i = 3,4,6,8,10$$

$$\sum_{i=1}^{5} \frac{1^2}{z_6} = \left(\frac{1}{3}\right)^2 + \left(\frac{1}{4}\right)^2 + \left(\frac{1}{6}\right)^2 + \left(\frac{1}{8}\right)^2 + \left(\frac{1}{10}\right)^2 = 0.216$$

$$A_L = \frac{28{,}000\ (12.6)(4.64)}{3600 \times 11{,}100 \times 4} = 0.0102 \text{ ft}^2$$

$$A_L = 1.44 \text{ in}^2 \tag{39}$$

Correcting additionally by the reduced induced velocity through the slot to say 1/3 of that near the wall, the area should be increased by 3 times.

Therefore, the louver area should be

$$\underline{A_L = 4.32 \text{ in}^2} \tag{40}$$

Evaluation of Constants

$$u = - \frac{C_1^1}{z} \frac{C_2^1}{r} \tag{41}$$

$$v = \frac{800}{r^2} \tag{42}$$

$$w = C_1^1 \left[\frac{3}{r} \ln \left(\frac{z \tan \beta}{r}\right) - \frac{1}{r}\right] + C_2^1 \frac{z}{r^2} \tag{43}$$

at

$$z = 10 \text{ ft and } r = z \tan \beta$$

$u = u_1$ which is determined from the volumetric addition of air at that point. Also, the volumetric rate of flow Q may be determined from

$$Q_v = \int_0^{0.8} 2\pi r w \, dr \simeq \frac{34,000}{3600} \times 63.8 \frac{\text{lbm}}{\text{Hr}} \frac{\text{Hr}}{\text{sec}} \frac{\text{ft}^3}{\text{lbm}} \tag{44}$$

$$Q_v = \int_0^{0.8} 2\pi r \, w \, dr = 600 \frac{\text{ft}^3}{\text{sec}} \tag{45}$$

This yields two equations for the two unknown, C_1^1 and C_2^1.

This leads to a considerably lengthy evaluation and the following crude analysis is presented as a first cut for the separation capability.

The cross sectional flow is approximately through a cylinder of radius r and length of 10 feet. The density variation may be approximated by

$$\rho = C_4 r \tag{46}$$

where

$$C_4 = \frac{\rho}{r} = \frac{1}{\hat{v}_i r_i} = \frac{1}{(12.6)(3.2)} = 0.025$$

or

$$\rho = 0.025\ r \text{ or } \hat{v} = \frac{1}{0.025\ r} \tag{47}$$

At a radius of 1.0 ft., the specific volume is approximated as 40 ft^3/lbm which is fairly reasonable.

Then the average radially inward velocity may be determined by

$$u \simeq \frac{\dot{m}}{\rho A} = \frac{8.9\ \text{lbm/sec}}{(0.025r)(2\pi r)(10)}\ \frac{ft^3}{\text{lbm}}\ \frac{\quad}{ft^2}$$

or

$$u = \frac{5.68}{r^2}\ \frac{ft}{sec} \tag{48}$$

The particle velocity is approximately equal to the fluid velocity in the tangential direction which is

$$v = \frac{800}{r^2}\ \frac{ft}{sec}$$

Equation 19 then becomes

$$C_D \frac{\rho}{2} \left(\frac{5.68}{r^2}\right)^2 = \frac{4}{3}\ \frac{a\ (\rho_s - \rho)}{r} \left(\frac{800}{r^2}\right)^2 \tag{49}$$

or

$$\frac{24.0 \times r^2 \times \mu \times \rho \times (5.68)^2}{2\rho(5.68)\ a \times 2} = \frac{4}{3}\ a\ \frac{(\rho_s - 0.025r)}{r}\ (800)^2 \tag{50}$$

$$a^2\ (\rho_s - 0.025r) = \mu r^3\ (4.0 \times 10^5) \tag{51}$$

Now the viscosity for air is a function of temperature and pressure. Neglecting for the moment the pressure effects, the reduced temperature varies as $T_r \simeq \frac{Tr_o(r_o)}{r}$ approximately under the preceding assumptions.

For air

$$T_c = 132.0° \text{ K} = 238° \text{ R}$$

$$P_c = 36.4 \text{ ATM} \qquad \mu_c = 193 \times 10^{-6} \frac{\text{gm}}{\text{cm sec}}$$

according to Bird, Lightfoot, and Stewart (1).

For the low density limit, the reduced viscosity may be expressed approximately as

$$\mu_r = A' + B' \ln T_r \tag{52}$$

at

$$T_r = 1.0, \ \mu_r = 0.45$$

$$A' = 0.45$$

Then at

$$T_r = 5.0, \ \mu_r \simeq 1.75$$

$$B' = \frac{(\mu_r - A')}{\ln T_r} = \frac{1.75 - 0.45}{\ln 5.0} = \frac{1.30}{1.61} = 0.8$$

So that

$$\mu_r \simeq 0.45 + 0.8 \ln T_r \tag{53}$$

or

$$\mu = \mu_c \left[0.45 + 0.8 \ln \left(\frac{7.1}{r}\right)\right] \tag{54}$$

or

$$\mu = 12.9 \times 10^{-6} \left[0.45 + 0.8 \ln \frac{7.1}{r}\right] \frac{\text{lbm}}{\text{ft-sec}} \tag{55}$$

Thus

$$a^2(\rho_s - 0.025r) = r^3(5.15 \times 10^{-10}) \left[0.45 + 0.8 \ln \frac{7.1}{r}\right] \tag{56}$$

This may then be solved for the equilbrium radius for various sized particles.

$$a^2 = \frac{r^3(5.15 \times 10^{-10}) \left[0.45 + 0.8 \ln \frac{7.1}{r}\right]}{(\rho_s - 0.025r)} \tag{57}$$

Equation 57 was then solved for several values of radius and density values of the spheres. These results are presented as Table 1 along with values of the Reynolds number for these conditions which serves to validate the use of $C_D = \frac{24}{R_e}$.

III. Conclusions and Recommendations

Preliminary studies indicate that it is technically feasible to build a vortex incinerator for use in the burning of peanut hulls. Many of the operational problems have not been considered in this analysis.

It is recommended that further work be done in order to establish the operational characteristics in order to ascertain the power requirements and economic feasibility for such a device.

Density $\frac{lbm}{ft^3}$	Radius ft	Particle Size Microns	Reynolds Number
P_s	R	A	Re
32.00	3.20	7.32168	0.175
32.00	2.80	6.27914	0.160
32.00	2.40	5.23298	0.142
32.00	2.00	4.19475	0.122
32.00	1.60	3.17881	0.100
64.00	3.20	5.17397	0.124
64.00	2.80	4.43760	0.113
64.00	2.40	3.69854	0.100
64.00	2.00	2.96498	0.086
64.00	1.60	2.24706	0.070
96.00	3.20	4.22365	0.101
96.00	2.80	3.62262	0.092
96.00	2.40	3.01938	0.082
96.00	2.00	2.42058	0.070
96.00	1.60	1.83452	0.057
128.00	3.20	3.65741	0.087
128.00	2.80	3.13700	0.079
128.00	2.40	2.61465	0.071
128.00	2.00	2.09615	0.061
128.00	1.60	1.58866	0.050

TABLE II. Equilibrium Particle Sizes for Various Particle Densities and Radial Positions in the Postulated Flow Field

NOMENCLATURE

Symbol	Description	Units
a	particle diameter	ft
A	area	ft^2
B or β	cone half angle	radians
C_D	drag coefficient	
C_i	arbitrary constants	appropriate
C_i^1	arbitrary constants	appropriate
C_p	specific heat at constant pressure	Btu/lbm°R
D	diameter	ft
g_o	gravitational constant	$\frac{\text{lbm ft}}{\text{lbf sec}^2}$
h	enthalpy	Btu/lbm
$\dot{M}$	mass rate of flow	lbm/hr
u	radial velocity	ft/sec
Q	heat of combination	Btu/lbm
Q_v	volumetric rate of flow	ft^3/hr
R or r	radius	ft
Re	Reynolds number	
T	temperature	°F or °R
v	tangential velocity	ft/sec
$\hat{v}$	specific volume	ft^3/lbm
V	velocity	ft/sec

NOMENCLATURE (Con't)

Symbol	Description	Units
w	axial velocity	ft/sec
z	axial direction	ft
α	angle of louver entry with vertical	radians
ρ	density	lbm/ft^3

Subscripts

Symbol	Description
A	air
F	fuel (peanut hulls)
F_G	flue gases
L	louver
MAX	maximum anticipated
s	spherical particle
p	pressure
1	entering conditions
2	leaving conditions

APPENDIX B

APPROXIMATE ANALYSIS OF CENTRIFUGAL PRESSURE MAGNITUDE

Taking an element of rotating gas, at radius r, equating forces on inner and outer faces of the element and equating to the centripetal acceleration, one obtains

$$\frac{dp}{dr} = \frac{\rho w^2 r}{g} = \frac{\rho v^2}{rg}$$

Assuming Bagles law, (constant temperature)

$$P = k\rho$$

and substituting,

$$\frac{dp}{p} = \frac{w^2 r^2}{gk}$$

This may be readily integrated

$$-\ln P = \frac{w^2 r^2}{2gk}$$

or

$$P = P_o \, e^{\frac{w^2 r^2}{2gk}}$$

Assuming a nominal 0.3 psi ΔP across the exhaust places a nominal pressure P_o in the cyclone of 15 PSIA.

Assuming the outer circulation is circulation en-mass, i.e., little change in ω with r, $v = \omega r$, and $\omega = \frac{120}{r} = \frac{120}{0.54} = 222$, $\omega^2 = 49{,}284$, $k = 28{,}162$, $g = 32.2$

$$P = P_o \, e^{.02721}$$

and at r = 0.488 feet

$$P = 15 \, e^{.00648}$$

or

$$\approx 15.1 \text{ PSIA}$$

This corresponds to 2.71 inches of water.

Assuming $v \alpha \frac{1}{r}$ (instead of $v = \omega r$) the equation reduces to

$$P = P_o e^{\frac{\alpha^2}{2kg}\left[\frac{1}{R_o^2} - \frac{1}{R^2}\right]}$$

Using the measured value of ΔP @ $r = 0.488$,

$$\alpha = 17.75$$

$$P = P_o e^{0.0001737\left[\frac{1}{R_o^2} - \frac{1}{R^2}\right]}$$

Alternatively, if $v \alpha \frac{1}{r^2}$,

$$P = P_o e^{\frac{\alpha^2}{4kg}\left[\frac{1}{R_o^4} - \frac{1}{R^4}\right]}$$

Again using the measured value of ΔP @ $r = 0.488$

$$\alpha = 2.282$$

$$P = P_o e^{0.000001436\left[\frac{1}{R_o^4} - \frac{1}{R^4}\right]}$$

These values calculated at the radii of the ports are reported in Table III, and plotted in Figure 16.

TABLE III

THEORETICAL CENTRIFUGAL AND MEASURED STATIC PRESSURES

Radius Feet	Port No.	Measured in.H_2O	Theoretical Values (in H2O) (Assuming Velocity = f(r)) $v \alpha \frac{1}{r}$	$v \alpha \frac{1}{r^2}$	$v \alpha r$
0.533	Top				
0.488		3.8	3.8*	3.8*	2.71*
0.399		2.5	3.65	3.77	1.81
0.310		2.3	3.35	3.69	1.09
0.221		1.5	2.61	3.32	0.55
0.133	Exhaust				

*Value used to evaluate constants in equation.

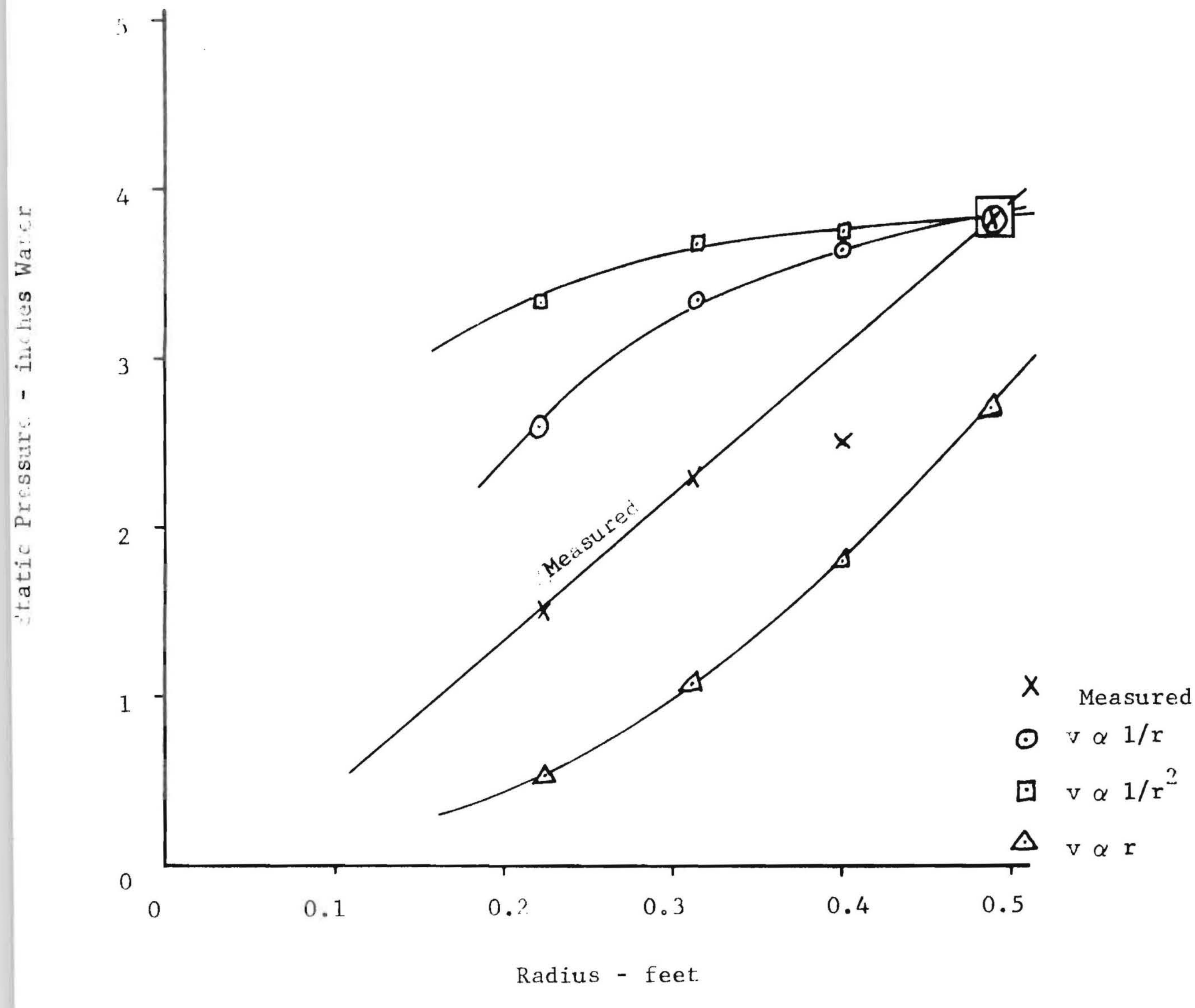

Figure 17. Comparison of Measured Static Pressure With Theoretical Centrifugal Pressure (assuming v = f(r))

APPENDIX C

RECENT PEANUT HULL INFORMATION FROM GEORGIA TECH

Some recent information has come to light from work done by M.D. Bower of Georgia Tech, and is reported below.

1. Preliminary DTA experiments (under direction of Dr. W.E. Moody)

Ignition of volatiles (in air) occurs in the range of 150°F - 350°F, and at this point degassing becomes exothermic, with interior 'hull' temperature in order of 800°F. Further heating results in carbonaceous, shell burning in manner similar to charcoal, with maximum interior 'hull' temperature 1200 - 1300°F.

2. Batch cooking of 'raw' peanut hulls in 30 lb. lots (in absence of other than residual air) suggests that once the hull temperature is up to 500 - 700°F, the 'coking' operation is exothermic. Although reasonable charcoal granules are obtained, a purer charcoal may be obtained by very limited external air at the 'near completion' point of the exothermic condition.

3. Adding additional oxygen (air) at this time (initially controlled to prevent chilling) will burn the charcoal to ashes - although nominal particle sizes of the peanut 'charcoal' suggest forced air burning times in minutes (1-3, probably).

This information suggests there may be at least two approaches to 'incineration' of light-weight waste (carbonaceous) material.

a) Complete burning to ash

b) Partial burning to a useful product (i.e., charcoal, particularly in case of peanut hulls).

At the present time, both processes appear feasible in the 'vortex-type' incinerator.